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SELECTION OF OIL COOLERS TO AVOID CONGEALING

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MEMORANDUM REPORT

for the

Army Air Forces, Materiel Command

and the

Bureau of Aeronautics, Navy Department

SELECTION OF OIL COOLERS TO AVOID CONGEALING

By Dennis J. Martin

SUMMARY

Many different oil coolers may be selected to dissipate any required amount of heat. The pressure drops and rates of air flow are readily determined from commercial data. However, there are three additional factors which are of vital importance:

1. Congealing tendency of the cooler
2. Power cost chargeable to the installation
3. Performance characteristics of the oil cooler in operation at altitude

The congealing tendencies of oil coolers rank in importance with pressure drop, power for cooling, and the dimensions of the unit. The congealing tendencies can be improved by selecting a unit of adequate size and then controlling the cooling by limiting the air flow.

This paper presents a method for (1) selecting an oil cooler which will dissipate the required amount of heat, (2) for determining its freezing tendency, (3) for calculating the power cost, and (4) for investigating the performance characteristics at any altitude.

INTRODUCTION

The selection and installation of an oil cooler to dissipate the required heat without congealing tendencies at an acceptable power cost has always been a troublesome problem. As the operating altitude and the velocity of airplanes have increased, the range of pressure drop available for cooling and the entrance air temperature have varied in such a manner that the congealing tendencies of oil coolers have been greatly aggravated. Also the increases in speed of new types of aircraft have put a premium on installations having optimum sizes of oil coolers. Careful seclection of an oil cooler is imperative to give satisfactory operation under all flight conditions.

The selection of the dimensions of any heat exchanger always involves a compromise among several quantities: proportions, power for cooling, pressure drop, etc. The problem of selecting an ethylene glycol or water radiator is one of finding the dimensions which will dissipate the heat with the pressure drop available over the altitude range of operation at the smallest power cost. Oil-cooler selection presents the additional problem of so selecting the dimensions and pressure drops that the cooler will be as free as practicable from congealing tendencies.

The present analysis is submitted in order to show what conditions must be satisfied to select an oil cooler which will give satisfactory operation under all flight conditions.

Appendices are included which present selection charts and selection forms for oil coolers.

SYMBOLS

A_a	total frontal area of cooler, square feet
A_o	area between baffles, perpendicular to oil flow, square feet
B	baffling constant determined from oil-cooler data
c_p	specific heat at constant pressure, Btu per pound per °F
c_1, c_2, c_4	empirical numerical constants
C_D/C_L	ratio of drag coefficient to lift coefficient of airplane
D_c	diameter of cooler, feet
D	hydraulic diameter of passage, feet
f	ratio of open area to total area
g	acceleration due to gravity, feet per second per second
h	coefficient of heat transfer, Btu per second per square foot per °F
H	rate of heat dissipation, Btu per second
H_p	heat transferred per 100° temperature difference between average oil and entering air
k	thermal conductivity, Btu per second per square foot per °F per foot
K_1	constant, seconds per foot-pound

L_a length of air passage, feet

n number of tubes per square foot of frontal area

p absolute pressure, pounds per square foot

Δp pressure drop, pounds per square foot

P power, foot-pounds per second

P_p pumping power, foot-pounds per second

P_w power to carry the oil cooler and supports (constant wing loading), foot-pounds per second

Q_a quantity of air flow, cubic feet per second

R ratio of the thermal resistance on the oil side to the thermal resistance on the air side

s effective cooling surface per unit length of tube, square feet per foot per tube

S effective cooling surface, square feet

T temperature, $^{\circ}F$

T_w temperature of tube wall, $^{\circ}F$

ΔT change in temperature, $^{\circ}F$

ΔT_i temperature difference between entering oil and inlet air, $^{\circ}F$

v volume, cubic feet

V_o speed of airplane, feet per second

W weight flow, pounds per second

u, v, x empirical numerical exponents

ϵ multiplying factor to account for weight of oil cooler mounting and supports

ζ mean effective temperature difference between oil and cooling air divided by ΔT_i

η_p duct pumping efficiency, useful power divided by drag power

μ coefficient of absolute viscosity, slugs per foot-second

η change in temperature of air divided by ΔT_i

ξ change in temperature of oil divided by ΔT_i

ρ density, slugs per cubic foot

ρ_w weight density of oil cooler, pounds per cubic foot

$\phi_1, \phi_a, \phi_o, \phi_1', \phi_a', \phi_o'$ constants

SUBSCRIPTS

a cooling air

i initial

o oil

t total

DEFINITION OF CONSTANTS

$$K_1 = \frac{n \Delta T_i}{H \phi_1 \phi_a}$$

$$\phi_1 = \frac{2 c_1}{D_a^{x+1} g^{3-x} \eta_p f_a^{2-x}} \times \frac{\mu_a^x}{\rho_a^2} = \left(\frac{\mu_a^x}{\rho_a^2} \times 10^3\right) \frac{\phi_1'}{\eta_p}$$

$$\phi_a = \frac{f_a^u D_a^{1-u} g^u}{s_a c_4} \times \frac{\mu_a^u}{k_a} = \frac{\mu_a^u}{k_a} \times \phi_a'$$

$$\phi_o = \frac{f_o^{\,v} D_o^{\,1-v} g^v B^v 2^v}{s_o\, c_3\, \pi^{v/2}} \times \frac{\mu_o^{\,v}}{k_o} = \frac{\mu_o^{\,v}}{k_o} \times \phi_o'$$

DEFINITION OF GENERALIZED VARIABLES

$$v' = K_1 \phi_1 \zeta v$$

$$A_a' = \frac{A_a}{W_a}$$

$$A_o' = \frac{\phi_o^{\,1/v}}{\phi_a} \frac{A_a \zeta L_a}{W_o}$$

$$P_p' = K_1 \zeta P_p$$

$$L_a' = K_1 \phi_1 \zeta W_a L_a$$

$$\Delta p_a = \frac{K_1 \zeta W_a \Delta p_a}{\rho_a g}$$

$$R = \frac{1/h_o S_o}{1/h_a S_a} = \frac{(A_o')^v}{(A_a')^u}$$

$$S' = \frac{A_a'}{(A_o')^2} \left(\frac{\phi_a}{\phi_o}\right)^{2/v} \frac{W_o^2}{W_a L_a^2}$$

ANALYSIS

Heat-transfer units, in general, involve two fluids and a dividing surface. When heat is being transferred from one fluid to another, there is a resistance to heat transfer $1/h_t S_t$. The total resistance to the flow of heat is the sum of the resistances of the two fluids and the dividing plate.

In most cases the dividing plate is very thin and its resistance may be neglected. For an oil cooler, we may then write

$$1/h_t S_t = 1/h_a S_a + 1/h_o S_o$$

In ethylene glycol or water radiators, the thermal resistance on the liquid side is negligible. In oil coolers, however, the thermal resistance on the liquid side $1/h_o S_o$ is much larger than in coolant radiators and under certain conditions may become equal to or even greater than the thermal resistance on the air side $1/h_a S_a$.

Because the physical properties of aviation lubricating oils, chiefly viscosity, depend very strongly on their temperatures, the prediction of oil-cooler performance is difficult. It is known that the oil tends to congeal near the tubes when the thermal resistance on the oil side is a large part of the total resistance, as occurs when the mass flow of oil is small, when the tube spacing is large, or when the oil side is not closely baffled. If the thermal resistance on the oil side is a small part of the total, the tube-wall temperature is near that of the oil at the center of the passage. The temperature gradient is similar to that shown in figure 1(a). As the thermal resistance on the oil side increases, the tube-wall temperature falls and the viscosity of the oil near the tube wall

increases rapidly as in figure 1(b). If the temperature of the tube wall decreases sufficiently the oil near the tube wall becomes very viscous and forms a layer which literally freezes onto the tubing. This reduces the effective hydraulic diameter of the passage and increases the oil pressure drop. This congealed layer acts as an insulating film which prevents the oil from performing its cooling function.

Figures 2, 3, 4, and 5 present typical test data taken at the Naval Aircraft Factory, Philadelphia. (See reference 1.) In figures 2 and 3 it will be noted that, as the cooling air temperature is reduced while the oil and air flows are maintained constant, an air temperature is ultimately reached where the heat dissipation decreases and the oil pressure drop increases until the bypass valve opens. In figure 4, the weight rate of flow of air is varied while the inlet air temperature and mass flow of oil are maintained constant. As the air flow is increased, a value is reached where the heat dissipation decreases and the oil pressure drop increases precipitantly. In figure 5, the heat dissipation is shown as a function of inlet oil temperature. In this case, as the inlet oil temperature is decreased a temperature is again reached where the heat dissipation begins to fall off.

The thermal resistance ratio, $\frac{1/h_o S_o}{1/h_a S_a}$, has been calculated for a number of points. From the approximate relation,

$$\frac{T_o - T_w}{T_w - T_a} = \frac{1/h_o S_o}{1/h_a S_a} ,$$

the tube-wall temperature at the point where heat transfer takes place between entering air and exit oil has been calculated and is plotted on the curves of figures 2, 3, 4, and 5. Similar analysis for a number of coolers indicates that congealing becomes possible for the grade of oil used when the tube wall falls below a temperature of approximately 100°F. A cooler should be selected with the tube-wall temperature always well above danger temperature.

It must be obvious that this criterion of congealing tendencies is much too simple to be used in a general case because other factors such as oil viscosity index, inlet oil temperature, and oil flow which were maintained in a limited range for the available test data are also important factors in determining the safe limiting value for the tube-wall temperature.

This analysis has been carried as far as possible without further information, where a large range of oil temperatures, oil flows, and viscosities has been used in heat-dissipation measurements on oil coolers. Such data might allow a correlation to be made which would permit a more general and more useful criterion of congealing tendencies.

By use of the appendix, figure 6 has been prepared for an oil cooler to dissipate 100 horsepower. Here the abscissa is weight flow of air while the ordinate refers to the various curves plotted. Curves showing pressure drop, power cost for cooling, frontal area, and tube-wall temperature are plotted on this figure. It is at once apparent from this figure that large oil coolers and small cooling air flow are favorable both for total power consumption and congealing tendencies. In order to see the picture of oil-cooler selection and operating characteristics with altitude, a set of three-dimensional charts has been prepared on the axes of frontal area and altitude of figures 7, 8, 9, and 10. Here the effect of altitude and frontal area on power consumption, weight flow of air, pressure drop, and tube-wall temperature is shown.

A study of these three-dimensional illustrations reveals several interesting points. As the altitude is increased with a given oil cooler, the weight flow of air must be reduced to dissipate the same amount of heat and to avoid congealing. If the air flow is held to the proper value to dissipate the heat, the maximum altitude is the critical altitude for the selection of an oil cooler with safe congealing tendencies. An oil cooler selected with a safe

tube-wall temperature at the maximum altitude should experience no congealing troubles at lower altitudes. For the example selected in the appendix, a cooler selected near minimum power has resonable freezing tendencies at altitude.

CONCLUSION

The results of this analysis of oil-cooler performance indicate that an oil cooler selected upon the basis of using the total pressure drop available will be small, will require excessive power expenditure, and may have very bad congealing tendencies. Although it might seem most logical to pick the smallest cooler which will dissipate the heat using what pressure drop there is available, in most cases the use of a larger cooler and a smaller pressure drop with good control over the air flow would consume less total power, would be less likely to congeal, and would require a smaller mass flow of air. The advantages of a larger oil cooler are realized only when careful control of the air flow is maintained.

As an example assume 70 pounds per square foot pressure drop is available in climb at 30,000 feet. An oil cooler using all the pressure drop available may be selected. From figure 9, A_a is found to be 0.82 square foot. From figures 7, 8, and 10: W_a = 5.7 pounds per second, T_w = 99°F, and P_t = 30 horsepower. This cooler would very likely congeal.

However, if only 30 pounds per square foot pressure drop is used, from figure 9, A_a = 1.16 square feet. Again from figures 7,8, and 10: W_a = 4.8 pounds per second, T_w = 111°F, and P_t = 14.2 horsepower. This arrangement would have much better freezing tendencies and would consume less than half the total power used by a cooler using all the pressure drop available.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., July 24, 1943.

APPENDIX I

DERIVATION OF THE OIL-COOLER EQUATIONS

In reference 2, a generalized selection chart for coolant radiators was presented. In reference 3, generalized equations for selection charts for heat exchangers in aircraft were derived. Selection charts for any type of heat exchanger can be constructed from these equations. In reference 4, generalized selection charts for air-to-air intercoolers were constructed. In this paper a generalized selection chart for oil coolers is presented. A generalized selection chart is valuable in that, since it gives a picture of the relations among the variables, it enables one to effect satisfactory compromises. A generalized chart approaches the ultimate in correlation. A single chart applies to all heat exchangers of a given internal design.

In both the construction and the use of an oil-cooler selection chart, the situation is quite different from that for coolant radiators or for intercoolers. For the correlation in coolant radiators, the liquid is in turbulent flow and it is not necessary to take into account the variation of the physical properties or the velocity of the liquid. Only the air side need be considered and the correlation equations of reference 5, can be used. In oil coolers, however, the oil flow is the laminar rather that the

turbulent type and the physical properties and the velocity of the oil in the passages must be considered.

The power chargeable to the oil-cooler installation is the sum of the power required to pump the cooling air through the cooler plus the power required to carry the oil cooler and its supports,

$$P_t = P_p + P_w \tag{1}$$

Since the power consumption is not the critical variable, the weight-carrying power will be omitted in the selection chart and equations. This greatly simplifies the charts and reduces the computations. However, the weight-carrying power will be added before the final selection is made.

$$P_p = \frac{Q_a \, \Delta \, p_a}{\eta_p} = \frac{W_a \, \Delta \, p_a}{\rho_a \, g \, \eta_p} \tag{2}$$

The pressure drop for air flowing through tubes is given by

$$\frac{\Delta \, p_a}{4 \times \frac{1}{2}\rho_a V_a{}^2 L_a/D_a} = c_1\left(\frac{\mu_a}{\rho_a V_a D_a}\right)^x$$

or,

$$\Delta \, p_a = \frac{2 \, c_1 \, L_a \, \mu_a{}^x}{D_a{}^{x+1} \, g_a{}^{2-x} f_a{}^{2-x} \rho_a} \left(\frac{W_a}{A_a}\right)^{2-x}$$

hence, from equation (2),

$$P_p = \phi_1 \, v\left(\frac{W_a}{A_a}\right)^{3-x} \tag{3}$$

The total thermal resistance equation is,

$$\frac{1}{h_t S_t} = \frac{1}{h_a S_a} + \frac{1}{h_o S_o} = \frac{1}{n\ v}\left(\frac{1}{h_a s_a} + \frac{1}{h_o s_o}\right) \qquad (4)$$

The generally accepted equations for correlating heat-transfer coefficients are,

$$\frac{h_a D_a}{k_a} = c_3 \left(\frac{\rho_a V_a D_a}{\mu_a}\right)^u \qquad (5)$$

$$\frac{h_o D_o}{k_o} = c_4 \left(\frac{\rho_o V_o D_o}{\mu_o}\right)^v \qquad (6)$$

The free area of the oil passage,

$$A_o = L_a D_c f_o B = B\ f_o \sqrt{\frac{4}{\pi}}\ L_a \sqrt{A_a}$$

From equations (4), (5), and (6),

$$\frac{1}{h_t S_t} = \frac{1}{n\ v}\left[\frac{D_a\ \mu_a{}^u}{s_a c_4 k_a \left(\dfrac{W_a D_a}{g A_a f_a}\right)^u} + \frac{D_o\ \mu_o{}^v}{s_o c_3 k_o \left(\dfrac{W_o D_o}{g f_o B\ \sqrt{A_a} L_a 2/\sqrt{\pi}}\right)^v}\right] \qquad (7)$$

The heat-balance equation is,

$$H = h_t S_t \Delta T_i \zeta$$

Then from equation (7)

$$\frac{n\ v\ \Delta\ T_i\ \zeta}{H} = \phi_a \left(\frac{A_a}{W_a}\right)^u + \phi_o \left(\frac{\sqrt{A_a} L_a}{W_o}\right)^v \qquad (8)$$

To arrive at the generalized equations we define the following generalized variables:

$$A_a' = A_a/W_a$$

$$A_o' = \left(\phi_o/\phi_a\right)^{1/v} \sqrt{A_a} L_a/W_o$$

$$v' = n \; \Delta T_i \; v \zeta /H \; \phi_a = K_1 \; \phi_1 \; v \zeta$$

$$P_p' = n \; \Delta T_i \; P_p \zeta /H \; \phi_1 \; \phi_a = K_1 \; P_p \zeta$$

$$\Delta p_a' = K_1 \zeta W_a \; \Delta \; p_a/g \; \rho_a$$

From equation (3),

$$P_p' = v'/ (A_a')^{3-x} \tag{9}$$

From equation (8)

$$v' = (A_a')^u + (A_o')^v \tag{10}$$

and,

$$L_a' = v'/A_a' = K_1 \; \phi_1 \; \zeta \; L_a W_a \tag{11}$$

The ratio of the resistance to heat flow on the oil side to the resistance on the air side is,

$$R = \frac{1/h_o S_o}{1/h_a S_a} \tag{12}$$

Solving the generalized equations, R is found,

$$R = (A_o')^v \; / \; (A_a')^u \tag{13}$$

A convenient variable S' is defined,

$$S' = \frac{A_a'}{(A_o')^2} = \left(\frac{\phi_a}{\phi_o}\right)^{2/v} \frac{W_o^{\;2}}{W_a \; L_a^{\;2}} \tag{14}$$

In order to apply equations (9), (10), (11), (13), and (14) to the plotting of a generalized oil-cooler selection chart, the values of the exponents x, u, and v must be determined. The quantities x and u are the exponents occurring in the friction factor and heat-transfer equations on the air side,

$$\frac{\Delta p_a}{4 \times \frac{1}{2} \rho_a V_a^2 L_a/D_a} = c_1 \left(\frac{\mu_a}{\rho_a V_a D_a}\right)^x$$

and,

$$\frac{h_a D_a}{k_a} = c_3 \left(\frac{\rho_a V_a D_a}{\mu_a}\right)^u$$

In oil coolers, the cooling air generally flows through the tubes and the oil around and across the tubes. For turbulent flow of air through tubes, x has been given as 0.2 and u as 0.8 in reference 5.

The quantity v is the exponent in the equation correlating the heat-transfer coefficient on the oil side,

$$\frac{h_o D_o}{k_o} = c_4 \left(\frac{\rho_o V_o D_o}{\mu_o}\right)^v$$

In figure 11 are shown typical commercial data obtained on a 13-inch oil cooler. From these data the heat-transfer conductance $h_o S_o$ has been plotted in figure 12 as a function of weight flow divided by the viscosity at the average oil temperature.

Sample calculations are given in appendix II. The heat-transfer coefficient on the oil side h_o is seen to vary as the 0.5 power of W_o/μ_o. The value for the exponent v has therefore been taken as 0.5 in plotting the selection chart of figures 13(a) and 13(b). If later data are obtained that would change the value of this exponent, new charts could be constructed by the same method.

Figure 14 has been plotted from Nusselt's results for cross flow (reference 6) and gives ζ as a function of ξ for various values of η. The mean temperature difference for both counterflow and parallel flow is very nearly the same as for cross flow. Figure 14 may therefore be used in making calculations for oil coolers baffled so that part of the oil path corresponds to parallel flow and part corresponds to counterflow as well as oil coolers with pure cross flow.

APPENDIX II

CALCULATION OF THE OIL HEAT-TRANSFER COEFFICIENT

From figure 11 the heat dissipation per 100° temperature difference between average oil and inlet air for a given oil flow and airflow is obtained. The three heat-balance equations,

$$H = W_a \, c_{pa} \, \Delta T_a \qquad (15)$$

$$H = W_o \, c_{po} \, \Delta T_o \qquad (16)$$

$$H = h_t S_t \, \Delta T_i \, \zeta \qquad (17)$$

and the equations defining the thermal efficiencies,

$$\eta = \Delta T_a / \Delta T_i \qquad (18)$$

$$\xi = \Delta T_o / \Delta T_i \qquad (19)$$

$$\zeta = \Delta T_{effective} / \Delta T_i \qquad (20)$$

determine the over-all heat-transfer coefficient. Figure 14 may be used to determine ζ from η and ξ.

The over-all heat-transfer coefficient is related to the heat-transfer coefficient on the oil and air sides by the simple Ohm's law for thermal resistance,

$$1/h_t S_t = 1/h_a S_a + 1/h_o S_o \qquad (21)$$

The heat-transfer conductance on the air side is given by,

$$h_a S_a = S_a \, c_1 \, c_{pa} \, g \left(\frac{\mu_a}{D_a}\right)^{0.2} \left(\rho_a V_a\right)^{0.8} \qquad (22)$$

(Equation (22) may be derived from equation (5) assuming $k_a = 10\mu_a$)

$$c_1 = 0.0247$$

$$c_{pa} = 0.24 \text{ Btu/lb/}^{\circ}\text{F}$$

$$g = 32.3 \text{ ft/sec/sec}$$

The heat-transfer coefficient on the oil side therefore can be calculated.

The value of the constant ϕ_o' may then be found from the relation,

$$\phi_o' \; \frac{\mu_o{}^v}{k_o} \left(\frac{\sqrt{A_a}\; L_a}{W_o}\right)^v = \frac{1}{h_o S_o} \tag{23}$$

The following calculation will clarify the procedure. The data needed for this calculation are first listed. The dimensions of a typical 13-inch oil cooler are:

Diameter of cooler, inches	13
Length of cooler, inches	$9\frac{1}{2}$
Effective tube length, inches[a]	9
Frontal area of cooler, square foot	0.921
Volume, cubic foot	0.691
Number of tubes	1319
Outside diameter of tubes, inch	0.268
Inside tube diameter, inch	0.256
Distance across flats of hexagonal ends, inch	0.323
Total open frontal area, square foot	0.471
Total cooling surface, square feet	66.3

[a]There is a $\frac{1}{2}$-inch length of hexagonal tubing at each end of the $8\frac{1}{2}$-inch length of core tubing. The fin effectiveness of the hexagonal length is assumed to be 50 percent.

From figure 11, the heat dissipation of this cooler for an oil flow of 45 pounds per minute and an air flow of 250 pounds per minute is 1610 Btu per minute per 100°F temperature difference between average oil and inlet air temperature.

The inlet oil temperature is 225°F and the inlet air is 100°F. The arithmetic average of the oil temperature is

$$\overline{T}_o = T_{io} - \Delta T_o/2$$

The total heat dissipated is

$$H_t = H_p (\overline{T}_o - T_{ia})/100$$

From equation (15),

$$H_p (T_{io} - \Delta T_o/2 - T_{ia})/100 = W_o c_{po} \Delta T_o$$

solving for ΔT_o,

$$\Delta T_o = 65.9°F$$

The temperature difference between inlet oil and inlet air is $225° - 100° = 125°F$

$$\xi = 0.527$$

$$\eta = 0.197$$

From figure 14,

$$\zeta = 0.60$$

From equation (17),

$$1/h_t S_t = 3.04$$

From equation (22),

$$1/h_a S_a = 2.00$$

Equation (21) then gives.

$$1/h_o S_o = 1.04$$

The value of $\frac{\mu_o}{k_o}^V$ may be read from figure 17,

$$\frac{\mu_o}{k_o}^V = 1.24$$

Therefore, substituting in equation (23),

$$\phi_o' = 0.94$$

Before presenting an illustrative example, it is desirable to introduce the curves of figures 15, 16, and 17. Figure 15 shows $\frac{\mu_a^{0.2}}{\rho_a^2}$ as a function of average air temperature and pressure. The quantity ϕ_1 is given by the product of $\frac{\mu_a^{0.2}}{\rho_a^2}$ and a constant ϕ_1'; determined by the tube diameter D_a and the free area ratio f_a. Table I lists the values of ϕ_1' for various commercial-tube sizes.

In figure 16, $\frac{\mu_a^{0.8}}{k_a}$ is shown as a function of average air temperature. The quantity ϕ_a is given by the product of $\frac{\mu_a^{0.8}}{k_a}$ and a constant ϕ_a'; which is determined by the tube diameter and the free area ratio. Table I lists the values of ϕ_a' for various commercial-tube sizes.

In figure 17, $\frac{\mu_o^{0.5}}{k_o}$ is shown as a function of average oil temperature. The quantity ϕ_o is given by the product of $\frac{\mu_o^{0.5}}{k_o}$ and a constant ϕ_o'; which is determined by tube diameter, free area ratio, and the baffling design. The constant ϕ_o' may readily be determined from experimental data by the method given in appendix II.

APPENDIX III

ILLUSTRATIVE EXAMPLE

For a given internal design the selection of an oil cooler to dissipate a given amount of heat involves a choice among the following parameters: A_a, L_a, W_a, W_o, Δp_a, and R or T_w. For a given heat dissipation, the selection of three of these variables completely determines a cooler which will dissipate this required amount of heat. The engine manufacturer specifies the weight flow of oil, W_o. Oil coolers are commonly made in only 9-and 12-inch tube lengths. The 9-inch is usually found superior to the 12-inch in freezing tendencies, while the 12-inch may dissipate more heat for the same frontal area. Coolers may be selected for both 9-and 12-inch lengths and comparisons made. Thus, for a given length and a specified weight flow of oil, only one of the remaining variables may be chosen to completely determine the cooler.

The selection of pressure drop, Δp_a based on the pressure drop available, may not give a cooler with satisfactory congealing tendencies while the selection of a low value of the tube-wall temperature may require a cooler too large for the space available. Therefore, it is found desirable to select several coolers and arrange the values graphically so a satisfactory compromise may be made.

The selection chart and equations have been arranged in a convenient form. A selection of a value of weight flow of air W_a quickly determines all the remaining variables. Several values of W_a may be selected and the other variables plotted against weight flow of air as in figure 4. The simplicity of such a chart immediately becomes apparent. An oil cooler may now be selected, the compromise made with due regard to all the variables.

An oil cooler will now be calculated for sea-level operation with air at 60°F. The oil cooler is assumed to have the same geometrical arrangement, tube diameter, and the same internal baffle spacing as the 13-inch oil cooler described in appendix II. The design conditions are given in the following selection forms.

Selection Form I

(For all oil coolers)

Quantity	Symbol	Value	Units
Airplane velocity	V_o	469	ft/sec
Drag-lift ratio	C_D/C_L	0.125	
Weight factor	ϵ	1.5	
Duct efficiency	η_p	1.00	
Altitude		0	ft
Pressure at altitude		2116	lb/ft^2
Impact pressure	$\frac{1}{2}\rho V^2$	280	lb/ft^2
Inlet air pressure	p_{ia}	2396	lb/ft^2
Estimated pressure drop	Δp	20	lb/ft^2
Mean pressure	$\bar{p}_a$	2386	lb/ft^2
Temperature at altitude		59	^oF
Adiabatic temperature rise		18.4	^oF
Cooling air inlet temperature	T_{ia}	77.4	^oF
Weight flow of oil	W_o	2	lb/sec
Inlet oil temperature	T_{io}	225	^oF
Heat dissipation	H	70.7	Btu/sec
Oil temperature drop	ΔT_o	70.7	^oF
Average oil temperature	$\bar{T}_o$	190	^oF
Inlet temperature difference	ΔT_i	148	^oF
$\frac{\text{Oil temperature drop}}{\text{Inlet temperature difference}}$	ξ	0.478	

Oil-Cooler Selection Form 2

	Symbol	Value
Estimated weight flow of air	W_a	10
Air temperature rise	ΔT_a	30
Average air temperature	$\overline{T}_a$	92
$\frac{\text{Air temperature rise}}{\text{Inlet temperature difference}}$	η	0.199
From figure 14 at η and ξ	ζ	0.68
From figure 15(a) at $\overline{p}_a$ and $\overline{T}_a$	$\mu_a^{0.2}/\rho_a^2$	8.22×10^3
$1.926\ \mu_a^{0.2}/\rho_a^2 \times 10^{-3}$	ϕ_1	15.83
From figure 16 at $\overline{T}_a$	$\mu_a^{0.8}/k_a$	1.90
$3460\ \mu_a^{0.8}/k_a$	ϕ_a	6570
From figure 17 at T_o	$\mu_o^{0.8}/k_o$	1.28×10^3
$0.93\ \mu_a^{0.5}/k_o$	ϕ_o	11.95
$\frac{\eta\ \Delta\ T_i}{H\ \phi_1\ \phi_a}$	K_1	0.0288
$K_1\ \phi_1\ \zeta\ W_a\ L_a$	L_a'	2.152
	ϕ_a/ϕ_o	5.50
	$(\phi_a/\phi_o)^4$	915
$\left(\frac{\phi_a}{\phi_o}\right)^4 \frac{W_o^2}{W_a\ L_a^2}$	S'	650

Oil-Cooler Selection Form 3

At the intersection of the L_a' and S' curves on the generalized selection chart the values of the remaining generalized variables are read.

Generalized variable	Value of generalized variable	Constant	Value of constant	Variable	Value of variable
P_p'	50.0	$K_1 \zeta \times 550$	10.0	P_p	5.0 hp
A_a'	0.1747	$1/W_a$	0.10	A_a	1.75 ft^2
$\Delta p_a'$	50.0	$K_1 W_a/\rho_a g$	2.23	Δp_a	22.4 lb/ft^2
R	0.517			R	0.517

$$v = A_a L_a = 1.23 \text{ ft}^3$$

$$W = v \rho_w = 81 \text{ lb}$$

$$P_w = (C_D/C_L)V_o W/550 = 9.3 \text{ hp (Constant wing loading)}$$

$$P_t = P_w + P_p = 14.3 \text{ hp}$$

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5. McAdams, William H.: Heat Transmission. McGraw-Hill Book Co., Inc., 1933.

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TABLE I

Across hex	O. D.	I. D.	t	pitch	n	f_a	s_a	ϕ_1'	ϕ_a'
0.323	0.268	0.256	0.006	0.279	1594	0.569	0.0670	1.65×10^{-3}	3950
0.296	0.248	0.238	0.005	0.256	1898	0.586	0.0623	1.71	4280
0.272	0.230	0.220	0.005	0.236	2247	0.594	0.0576	1.76	4600
0.250	0.210	0.202	0.004	0.217	2660	0.592	0.0529	2.10	4910

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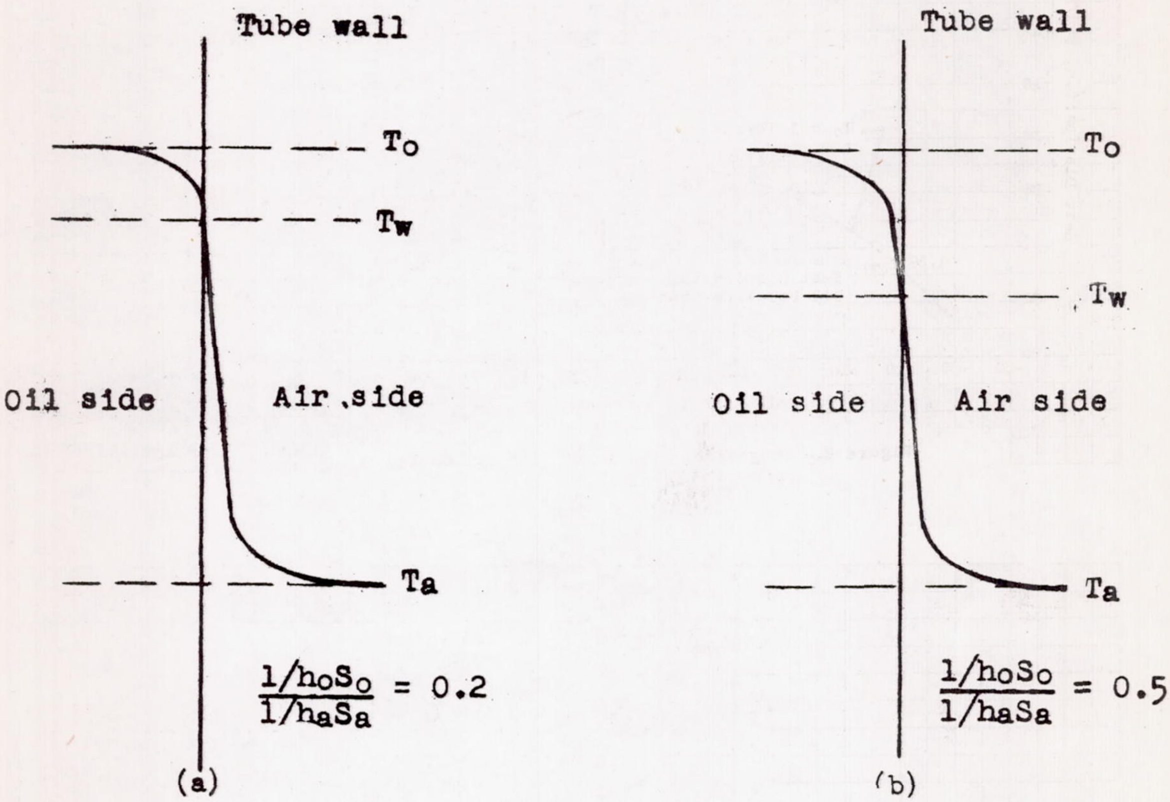

Figure 1.- Temperature gradient for oil to air heat transfer.

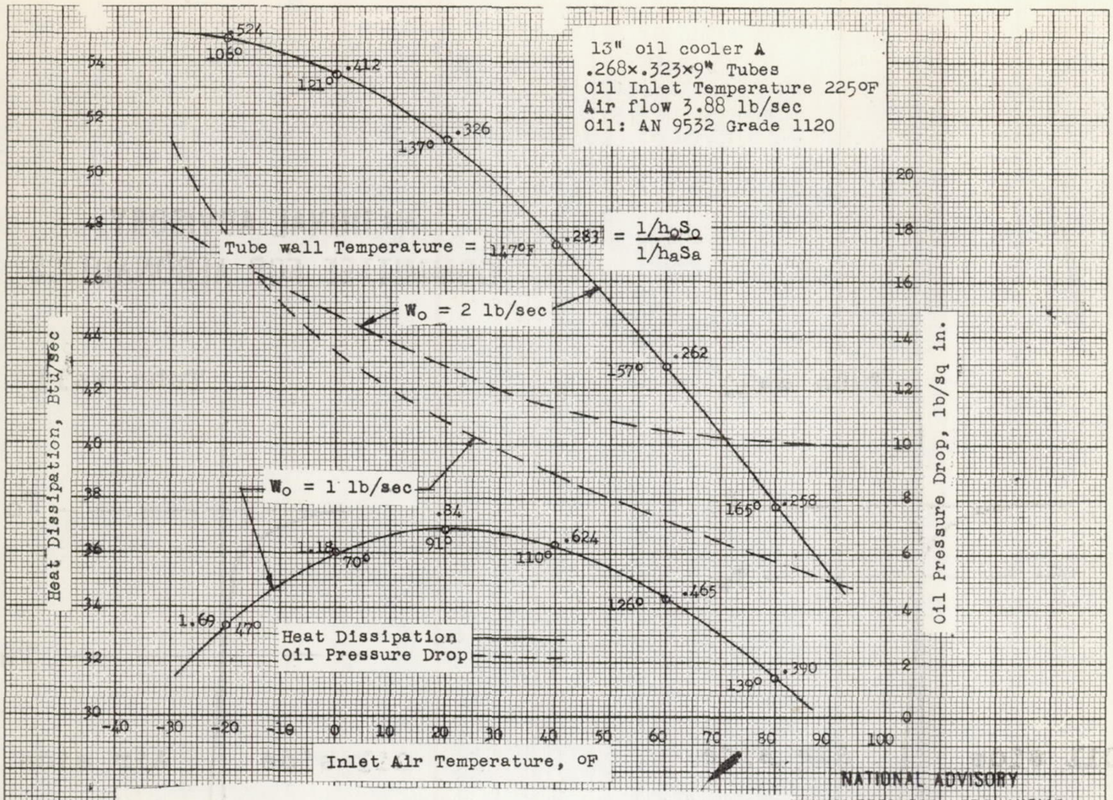

Figure 2.- Congealing tests of 13" oil cooler A.

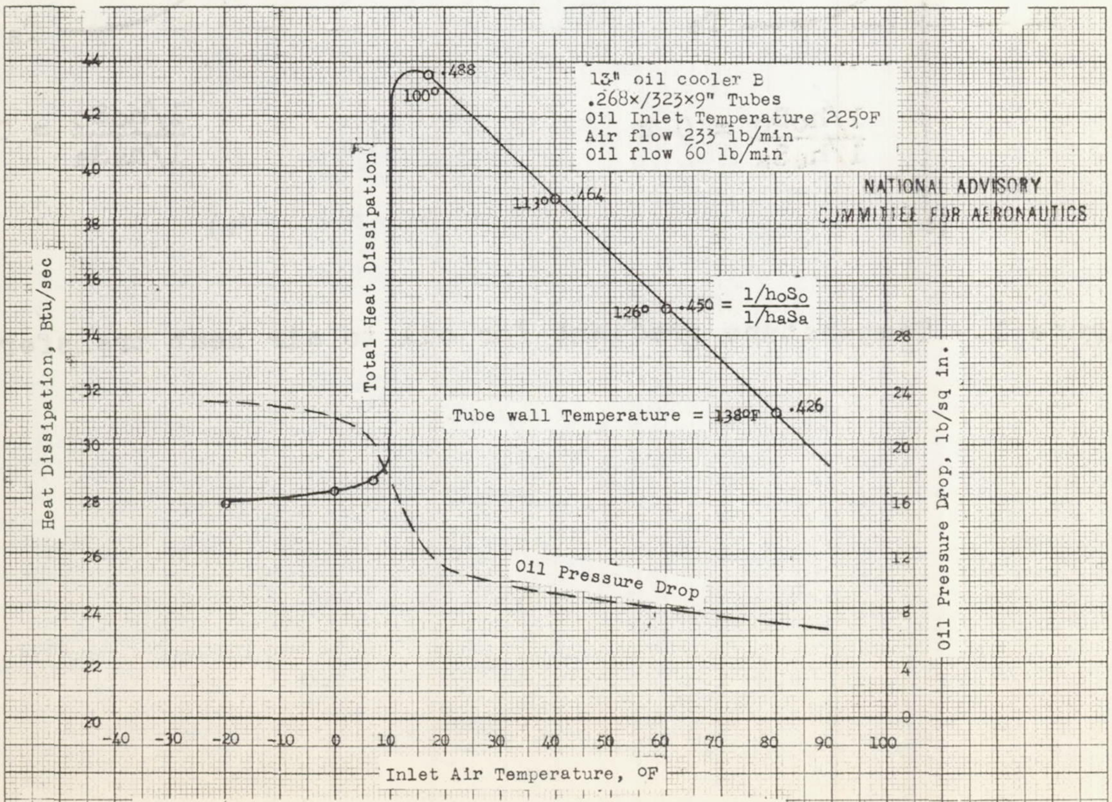

Figure 3.- Congealing tests of 13" oil cooler B.

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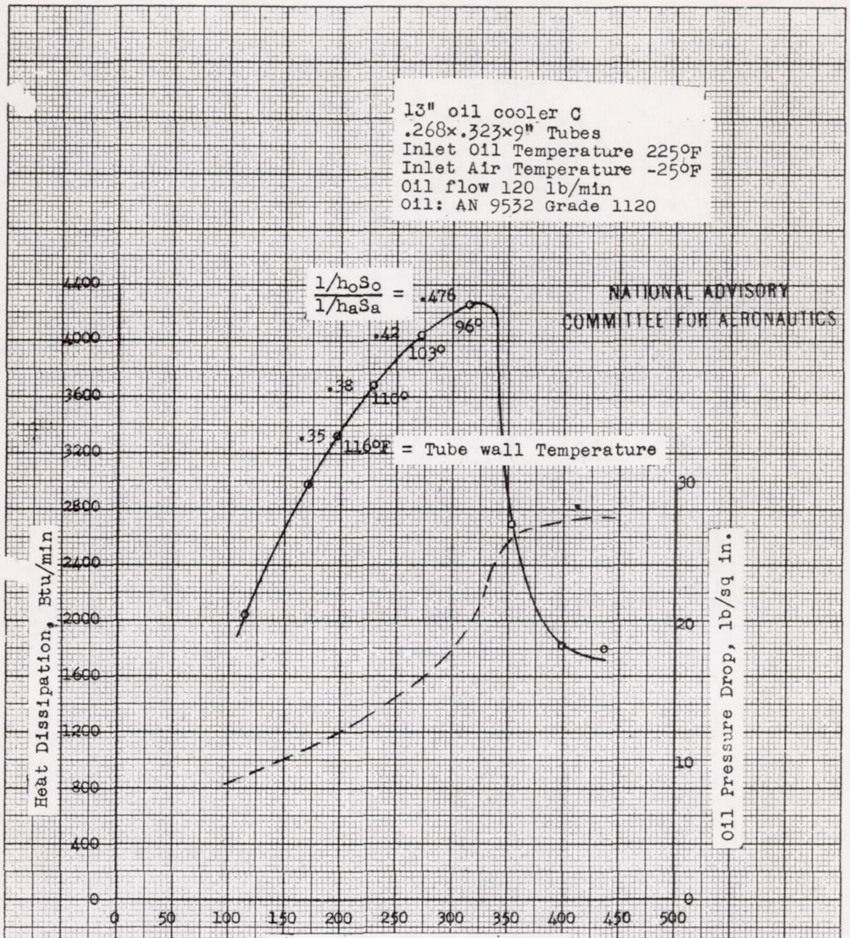

Figure 4.- Congealing tests of 13" oil cooler C.

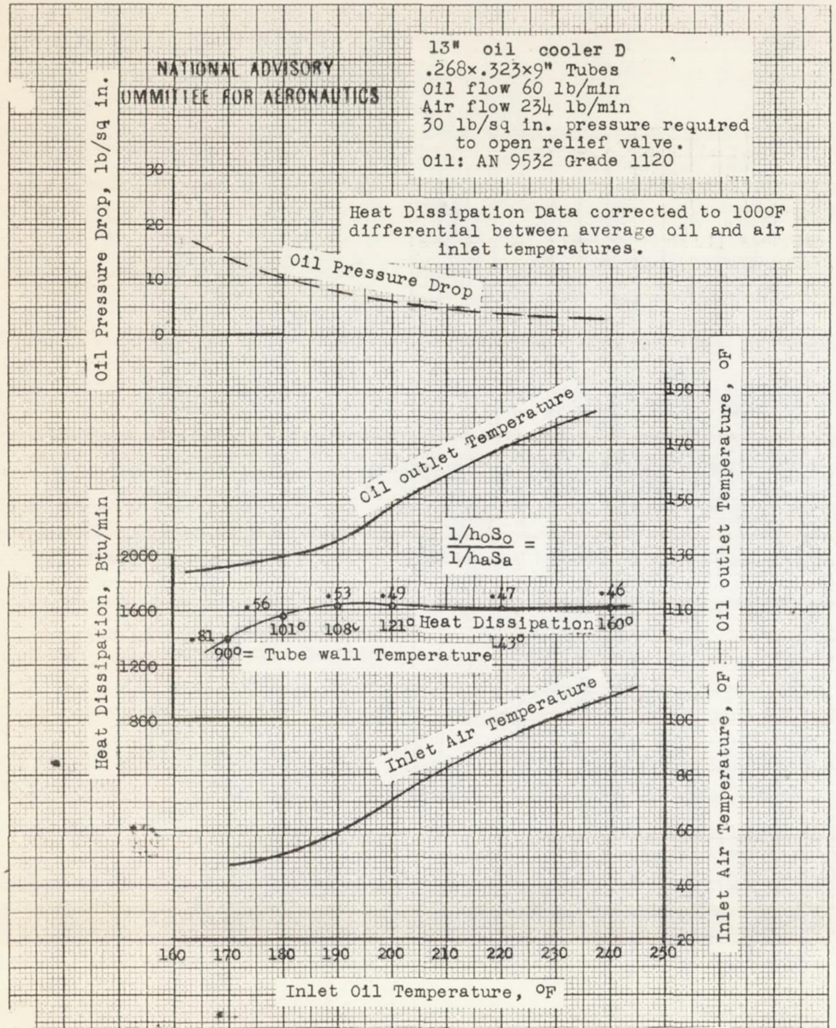

Figure 5.- Congealing tests of 13" oil cooler D.

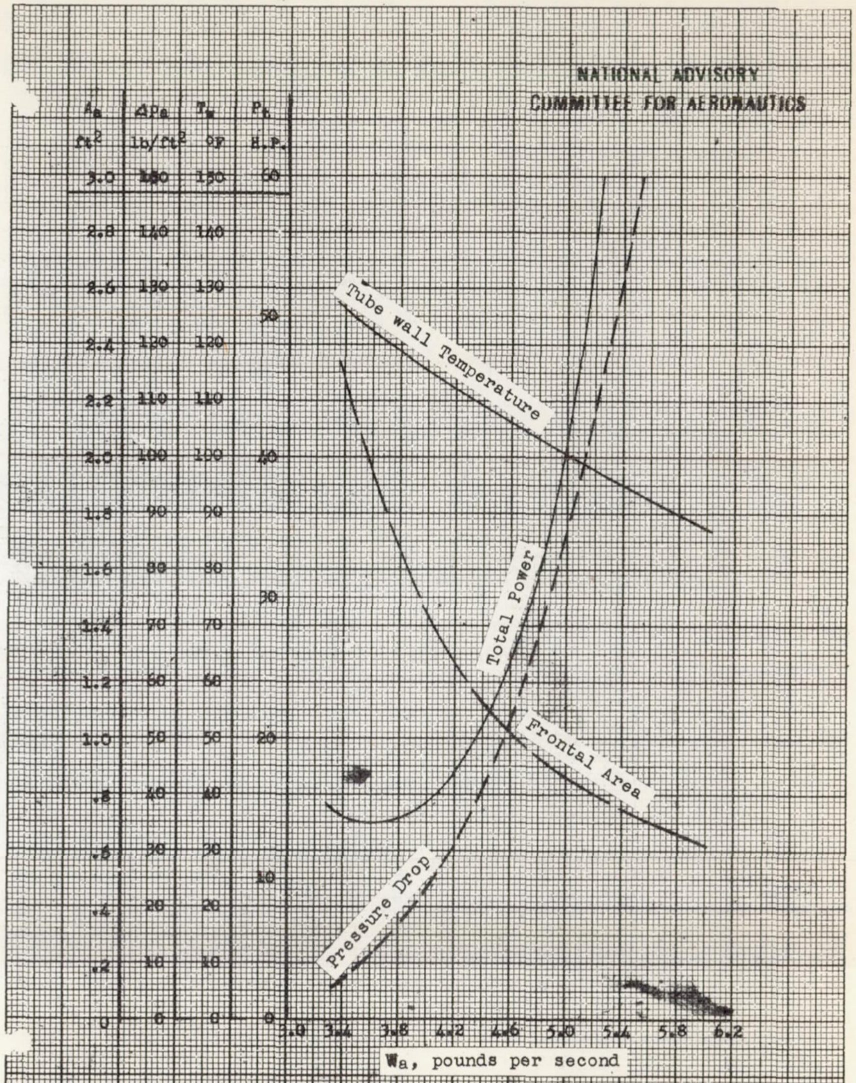

Figure 6.- Variation of oil cooler parameters with weight flow of air. Altitude, 40,000 feet.

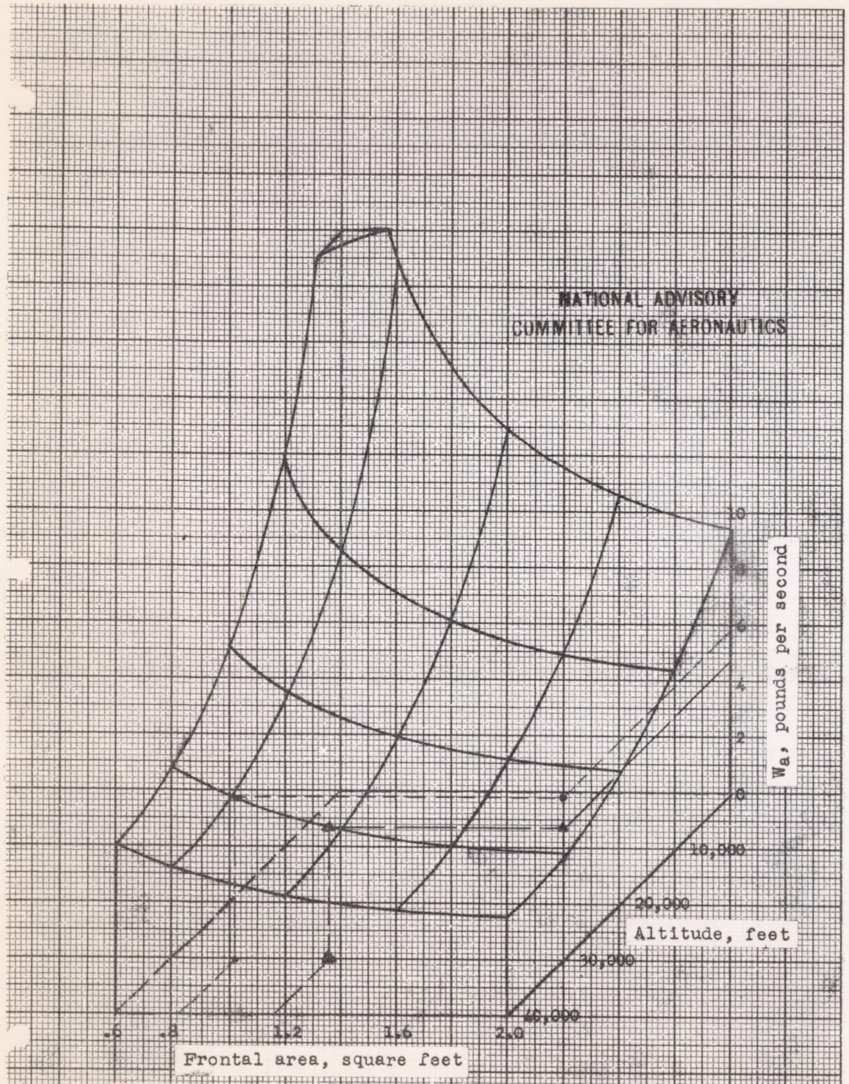

Figure 7.- Weight flow of cooling air as a function of frontal area and altitude.

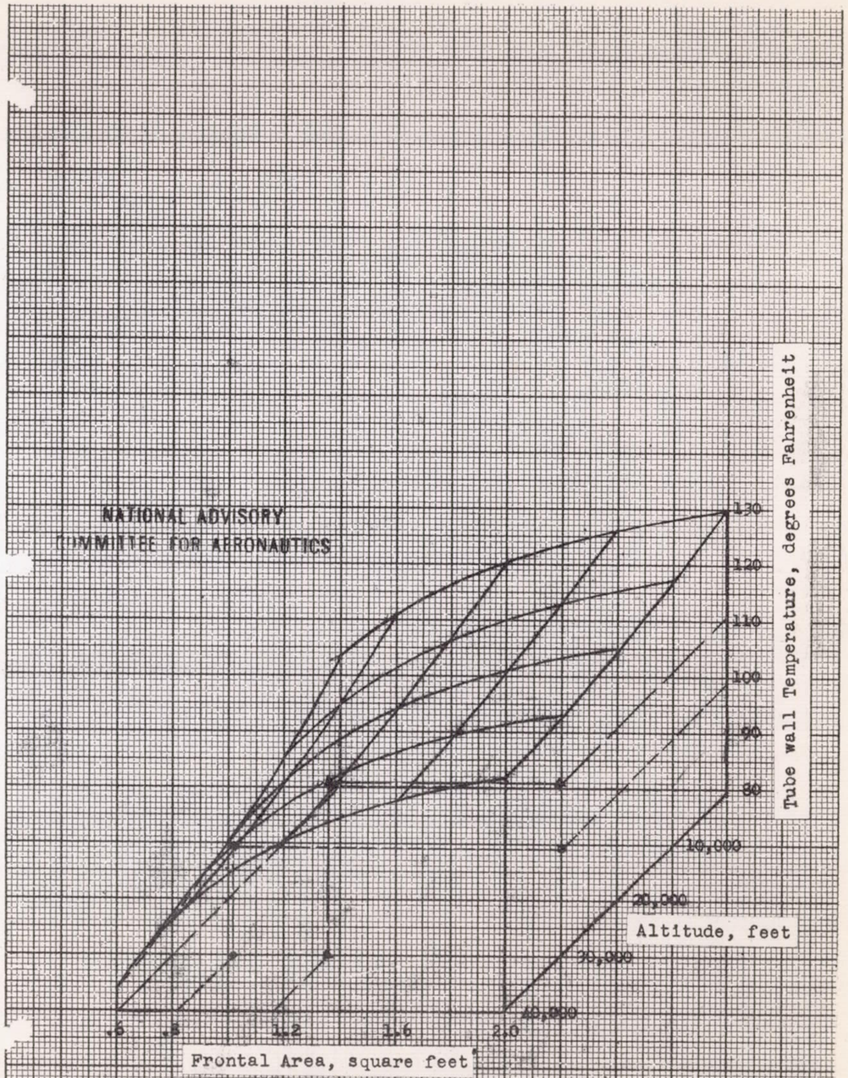

Figure 8.- Tube wall temperature as a function of frontal area and altitude.

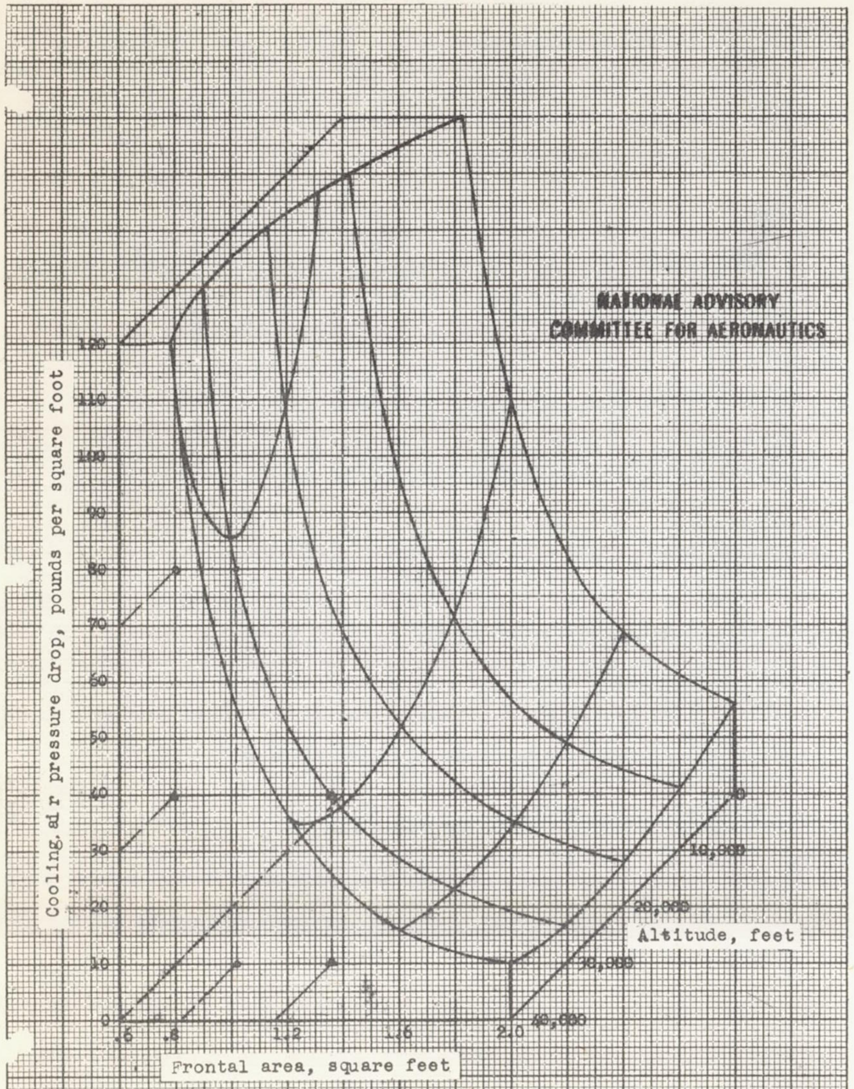

Figure 9.- Cooling air pressure drop as a function of frontal area and altitude.

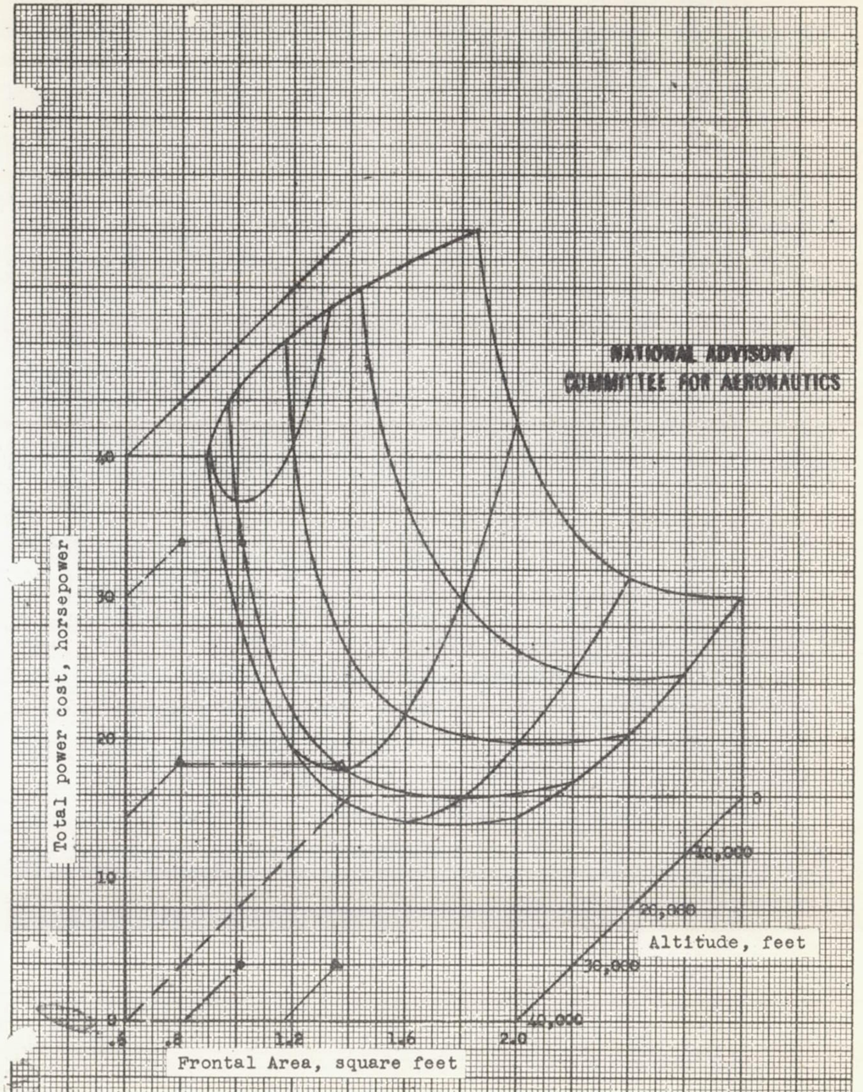

Figure 10.- Total power consumption as a function of frontal area and altitude.

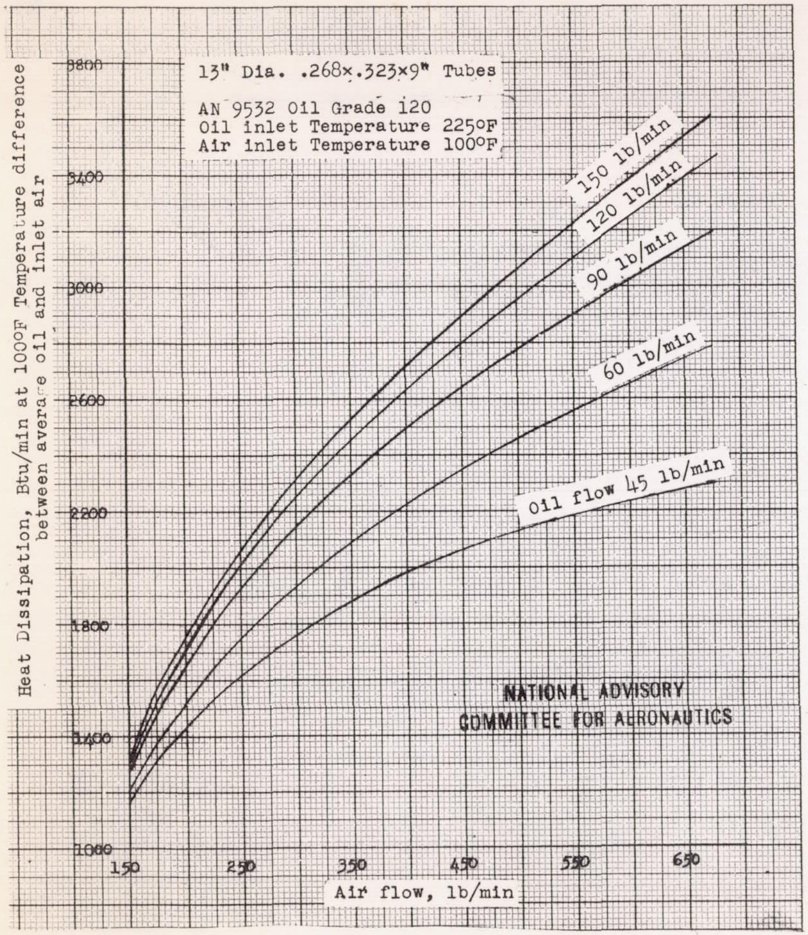

Figure 11.- Performance characteristics of 13" oil cooler .

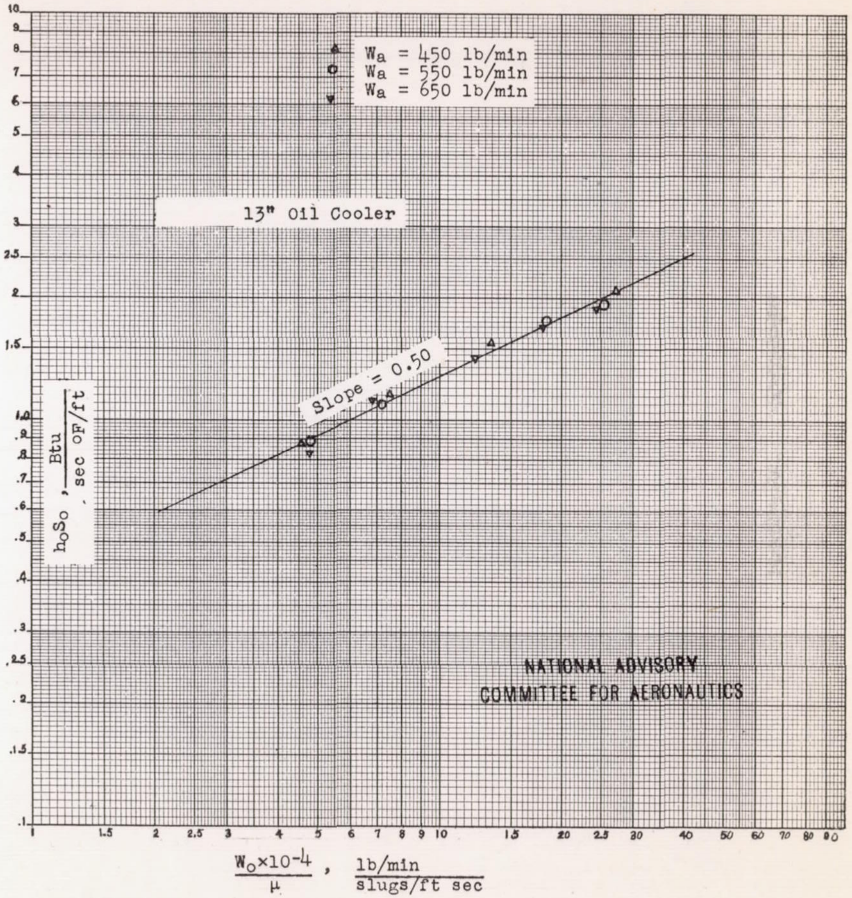

Figure 12.- Correlation of heat transfer coefficient on the oil side.

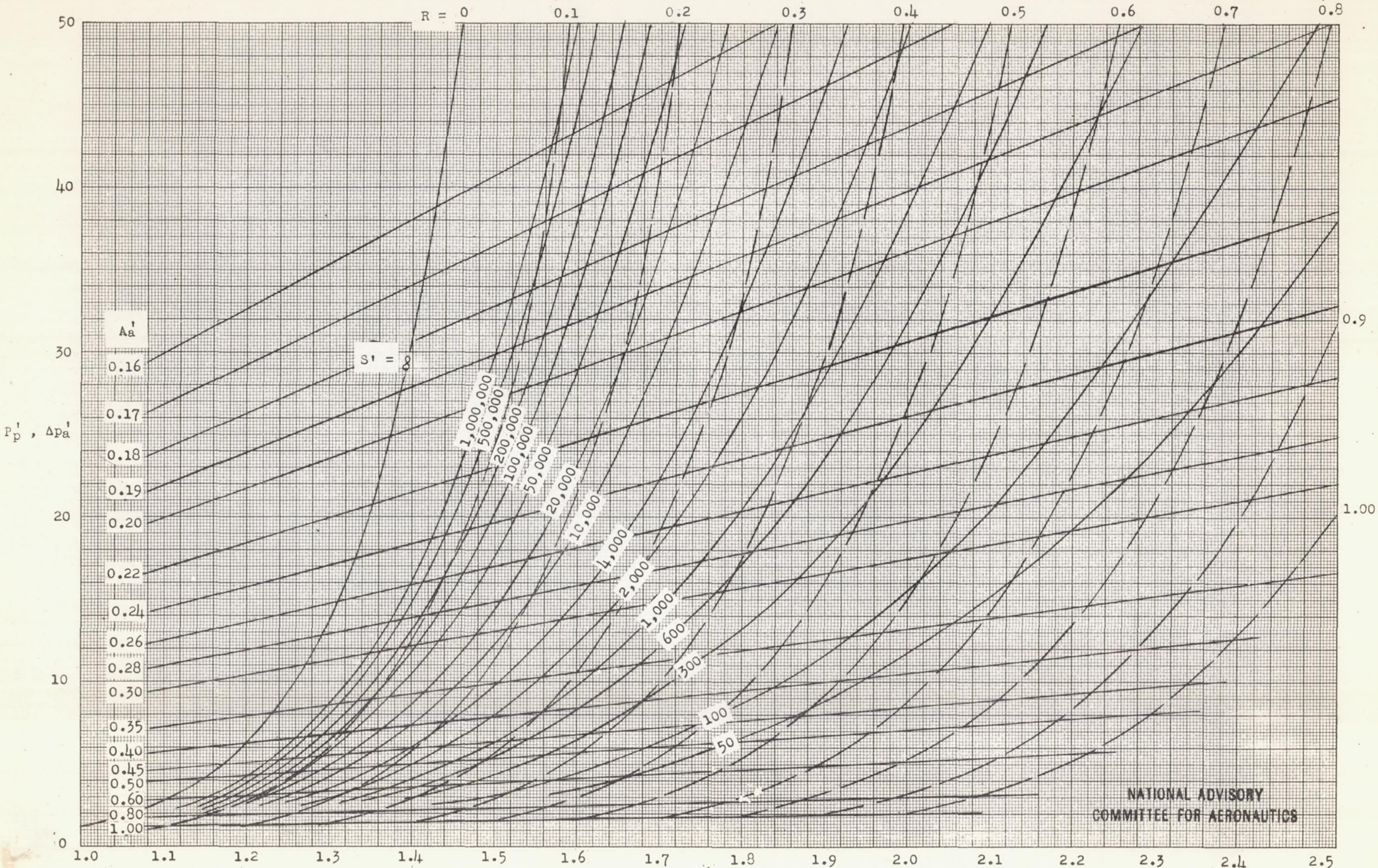

Figure 13.- Generalized Selection Chart for Oil Coolers.

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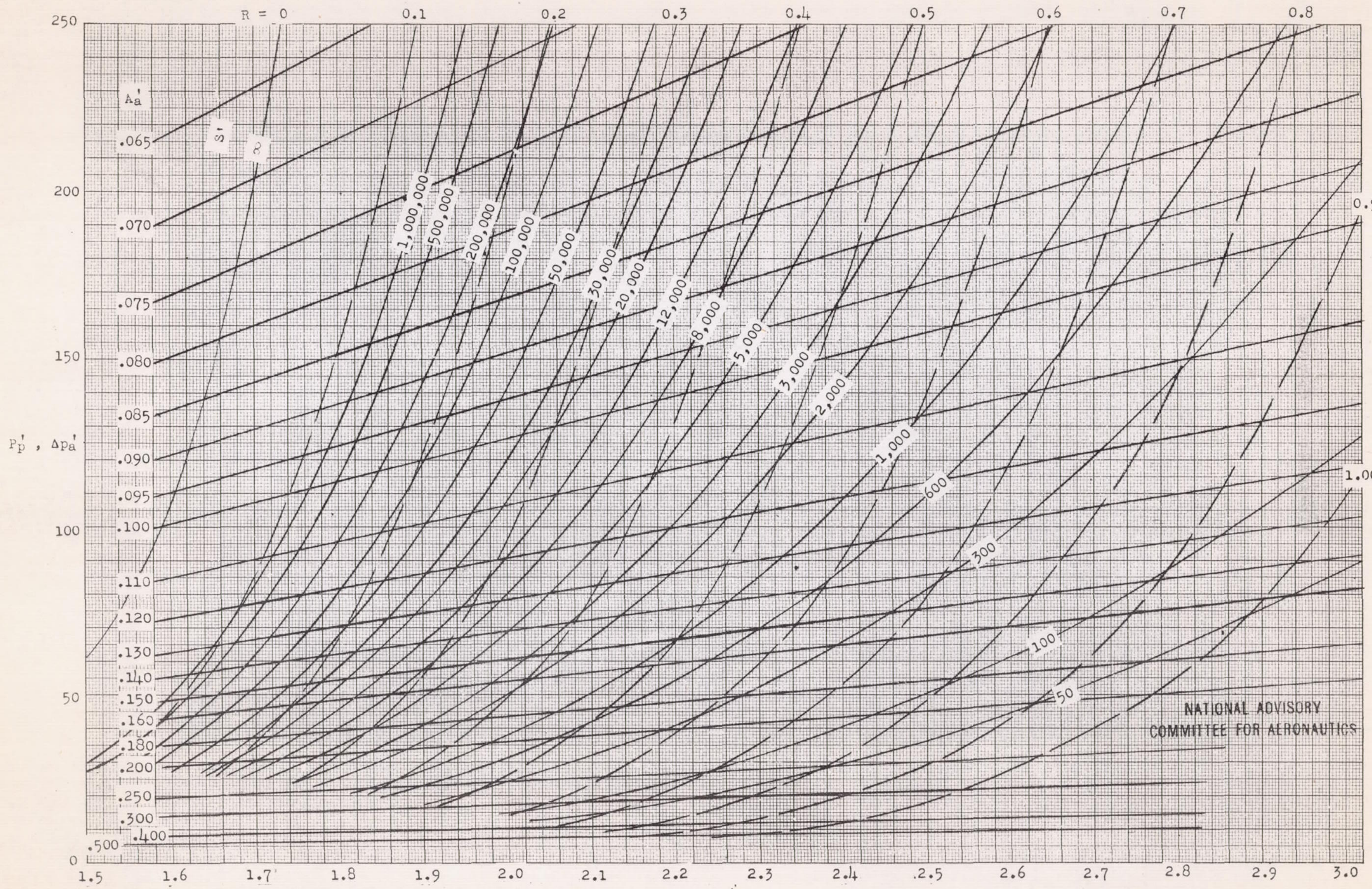

Figure 13.- Concluded, Generalized Selection Chart for Oil Coolers L_a'

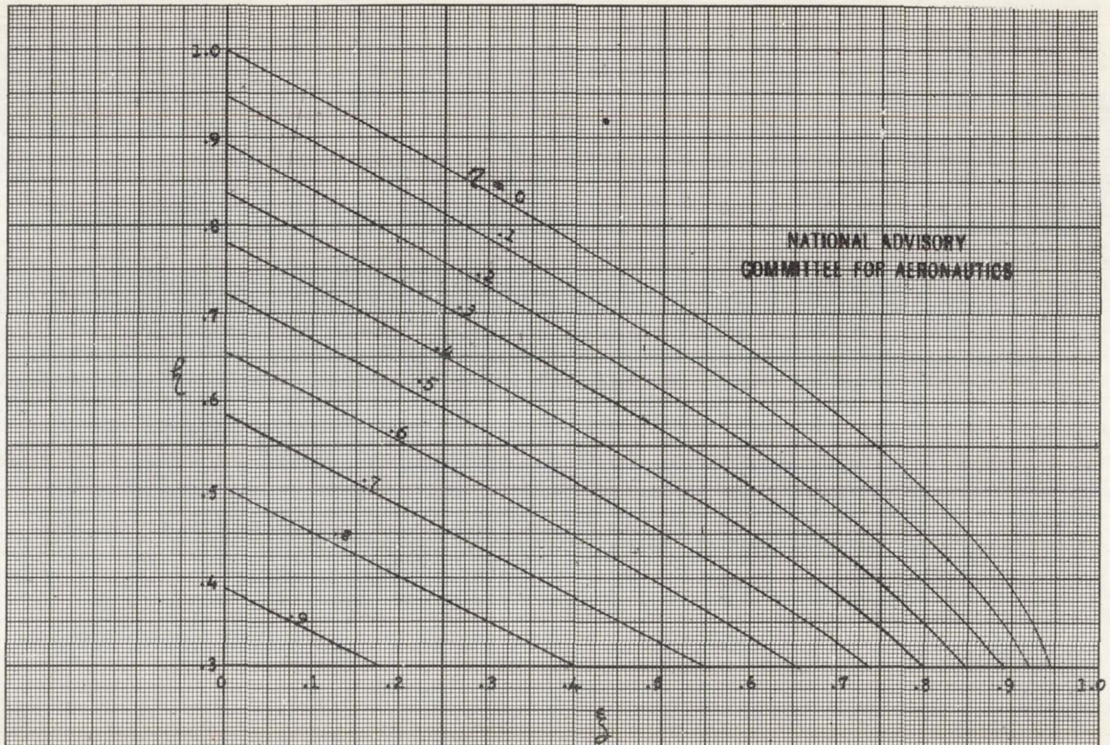

Figure 14.- ζ as a function of ξ and η.

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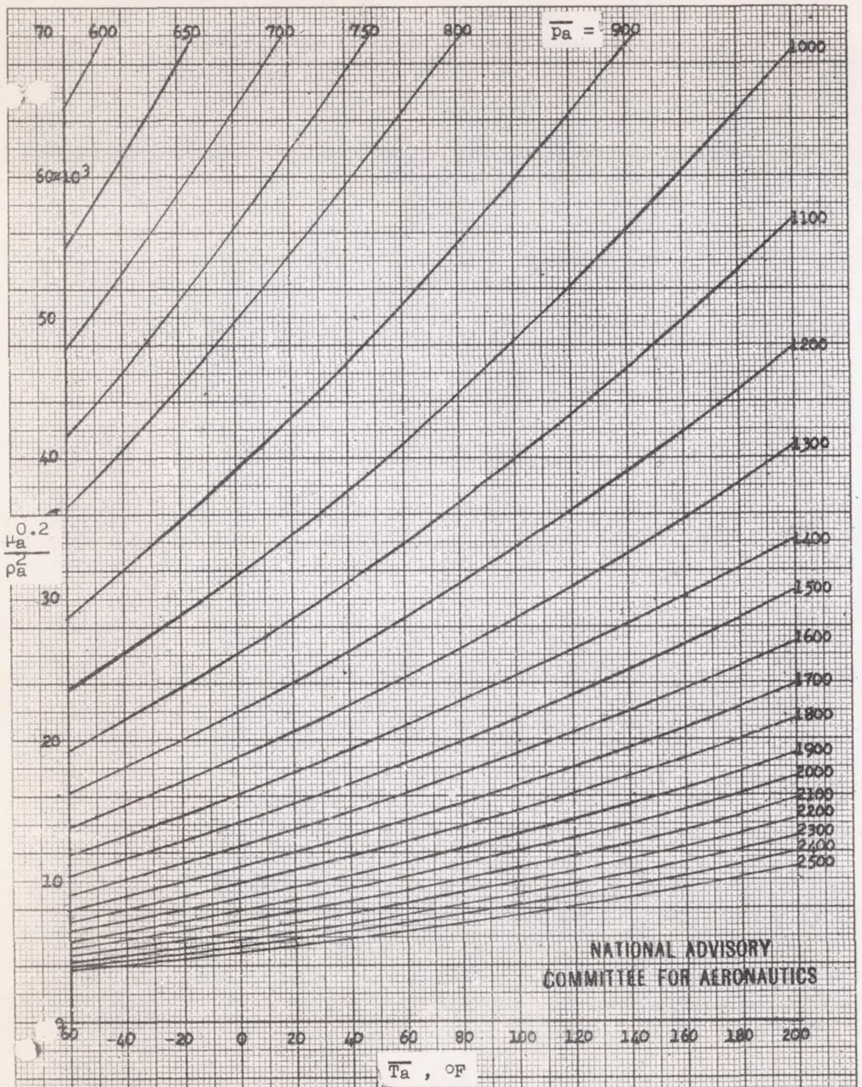

Figure 15.- $\frac{\mu_a^{0.2}}{\rho_a^2}$ as a function of $\overline{T_a}$ and $\overline{p_a}$.

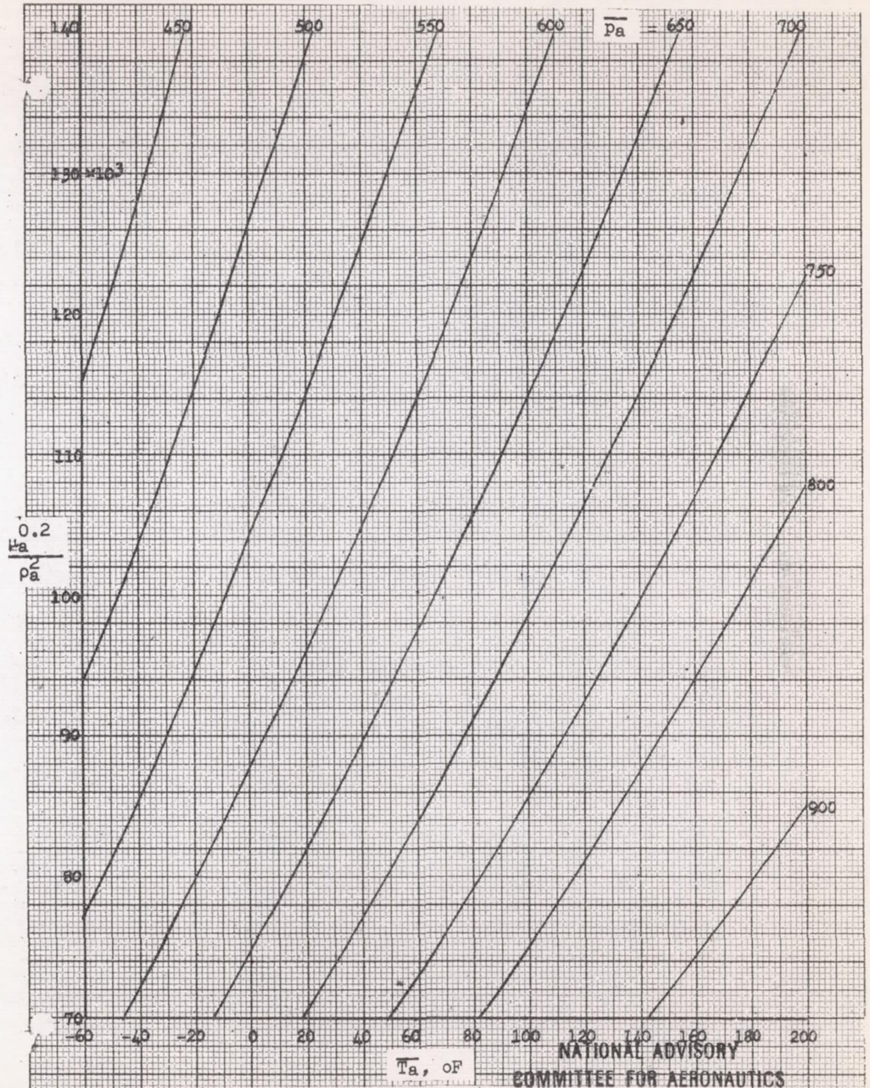

Figure 15.- Concluded, $\frac{\mu_a^{0.2}}{\rho_a^2}$ as a function of $\overline{T_a}$ and $\overline{p_a}$

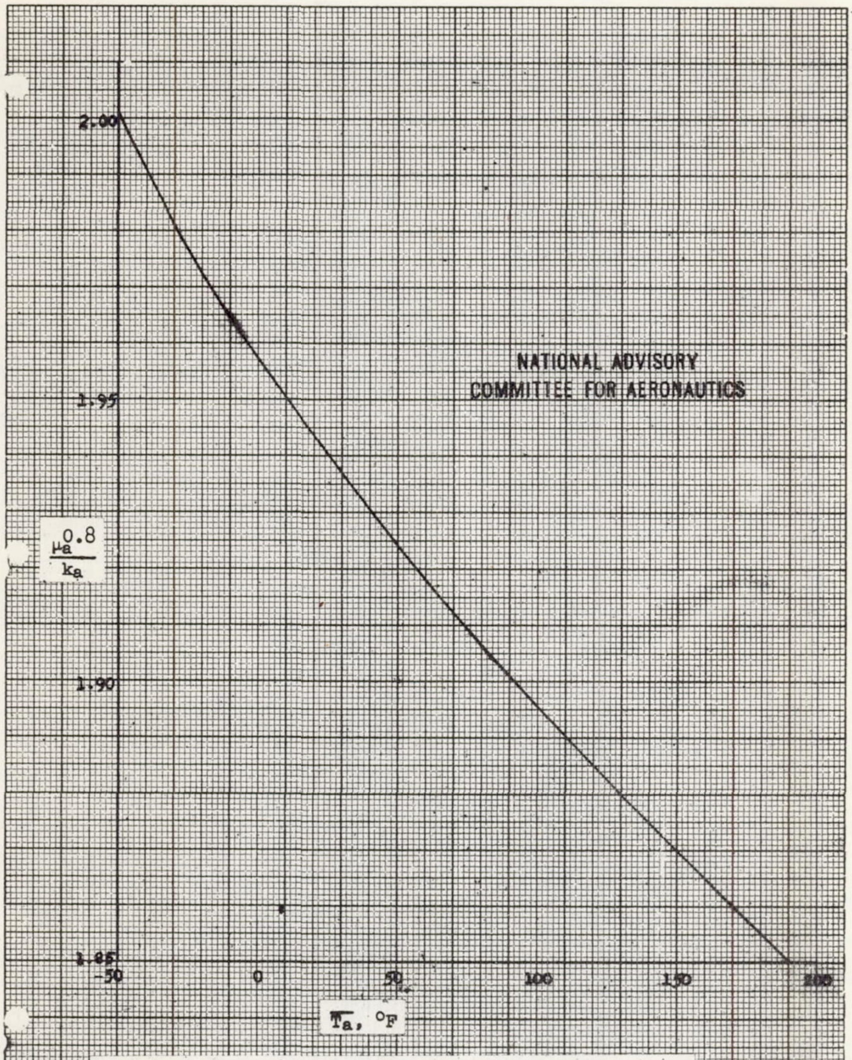

Figure 16.- The quantity, $\frac{\mu_a^{0.8}}{k_a}$ as a function of $\overline{T_a}$.

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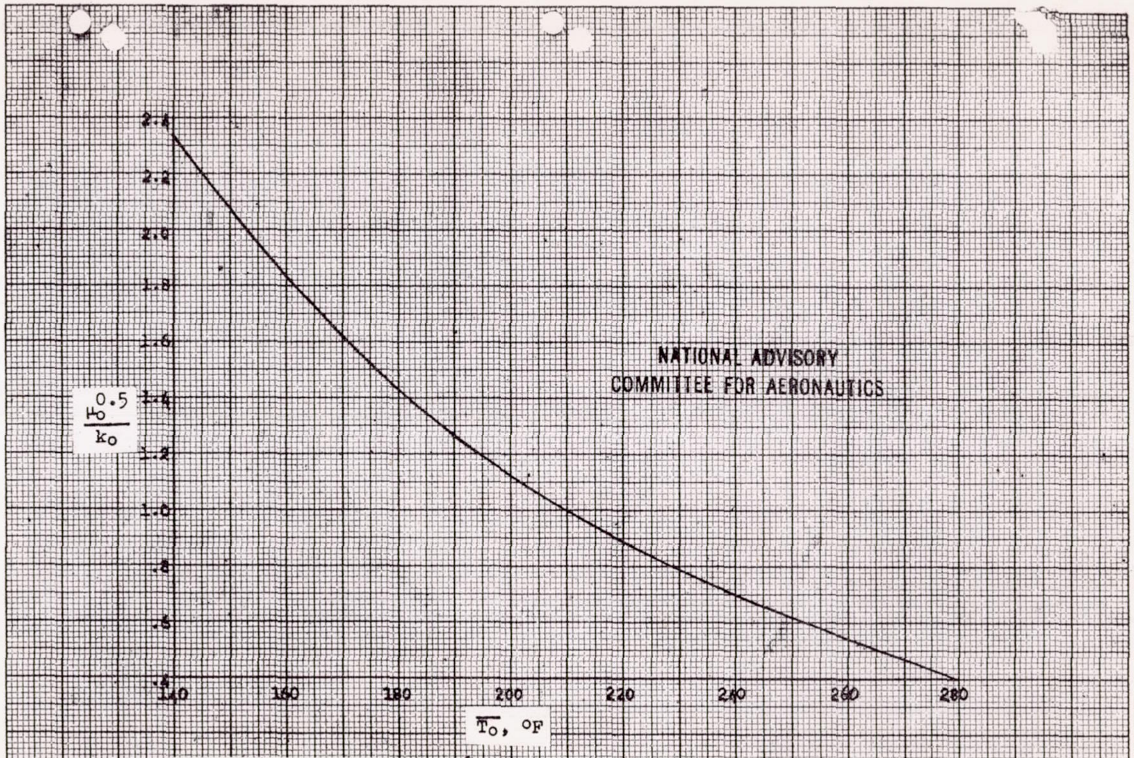

Figure 17.- The quantity, $\frac{\mu_o^{0.5}}{k_o}$ as a function of $\overline{T_o}$.